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# A mechanical transducer for low frequency sonic cleaning.

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# A MECHANICAL TRANSDUCER FOR LOW FREQUENCY SONIC CLEANING

Reynolds Beckwith

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A MECHANICAL TRANSDUCER

FOR

LOW FREQUENCY SONIC CLEANING

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Reynolds Beckwith

Letter on cover:

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FREQUENCY SONIC CLEANING

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# A MECHANICAL TRANSDUCER FOR

# LOW FREQUENCY SONIC CLEANING

by

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Submitted in partial fulfillment
of the requirements
for the degree of
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# PREFACE

In recent years sonic processing has become increasingly significant both in the laboratory and in industry. Applications include aiding chemical and physical reactions, soldering, drilling, and cleaning. The majority of these applications utilize frequencies in the ultrasonic rather than the audible region.

In order to clean objects completely by sonics without consideration for orientation to the transducer, it is necessary that the wavelength be long in relation to object size. Otherwise, shadow effect should shield these portions of the object away from the transducer. In the cleaning of metal parts, cavitation is the essential mechanism by which cleaning is accomplished.

It is the purpose of this paper to determine the practicability of a mechanically driven transducer for producing low frequency sound for cleaning purposes.

The experimental work was performed at the Sonics Division, Physics Department, Stanford Research Institute, Menlo Park, California.

The writer wishes to thank Professor L. E. Kinsler of the Naval Postgraduate School and Dr. Vincent Salmon of the Stanford Research Institute for their assistance, encouragement, and cooperation in the preparation of this paper.

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## LIST OF ILLUSTRATIONS

## TABLE OF SYMBOLS AND ABBREVIATIONS

(Listed in the order of their use in the text)

| | |
|---|---|
| $k$ | - The wavelength constant, $2\pi/\lambda$ |
| $a$ | - Radius |
| $\lambda$ | - Wavelength |
| $z$ | - Specific Acoustic Impedance |
| $\rho$ | - Density |
| $c$ | - Velocity of Sound |
| $\xi$ | - Displacement |
| $P$ | - Peak Sound Pressure |
| $t$ | - Thickness |
| $Y$ | - Young's Modulus |
| $\sigma$ | - Poisson's Ratio |
| $R_m$ | - Mechanical Resistance |
| $R_r$ | - Radiation Resistance |
| $X_r$ | - Radiation Reactance |
| $m$ | - Mass |
| $s$ | - Stiffness |
| $F$ | - Force |
| $A$ | - Acceleration |
| $P_{ave}$ | - Average Power |
| $S$ | - Area |
| $Z$ | - Acoustic Impedance |

# CHAPTER I

# INTRODUCTION

## 1. Sonic Processing.

Industrial use of sonic processing is composed of two major fields: small signal operation for inspection, testing, and measurement and high power operation in which sonic energy causes some chemical or physical change in the medium.

High power applications, presently being used in industry, include the following: treatment of leather, homogenization of milk, floatation of ore dressing, degassing of liquids, dispersion of dyes in the textile industry, dispersion in the processing of paints, better extraction of hops in the brewing process, the drilling of odd shaped holes in very hard materials, soldering of aluminum without flux, oil well drilling, micronization of slurries in paper processing, and the cleaning of small metal parts.

The cleaning of small metal parts is the application that is presently in the widest use in industry. The problem of cleaning electric razor heads has long been a difficult one for their manufacturers. Prior to sonic cleaning, the heads required hand brushing to remove the metal particles and lapping compound remaining after their manufacture. General Electric Corporation has installed a quartz transducer for the Schick Company and the Detrex Corporation has installed a fully conveyorized unit using barium titanate trough transducers for the Remington-Rand Company. Both of these units use solvent degreasing in addition to sonic energy in the cleaning process. The overall cleanliness of

the parts exceeds former standards.

Sonic energy is also being used to clean parts used in instruments. Bendix Aviation Corporation has developed a large magnetostrictive cleaning device for cleaning small parts used in their aviation instruments. The Japanese are using sonic cleaning for nozzles used in rayon spinning. In Germany, sonic cleaning devices are used to prevent the formation of precipitates during the production of glass parts for chemical use.

The Ultrakust Company of Germany has a domestic clothes washing machine on the market which utilizes an electrodynamic vibrator operating at twice line frequency (120 cps). What little information is available in the literature on this device, is not consistent in explaining its operation. One source claims the cleaning mechanism is not strictly dependent on cavitation, but relies on the high accelerations produced by the vibrating diaphragm to force streams of liquid through the clothes which loosen the soils. Another source states that the disc-shaped diaphragm is connected to the driver by a slender rod mounted in a tube through which an air stream is injected into the liquid load to decrease its mass reactance and to enhance gaseous-type cavitation.

## 2. THE CLEANING PROCESS

The earliest known method for cleaning metals was the use of abrasive powders, the most likely one being finely ground sand. Although Moser in 1842 published a paper indicating the desirability of ether and alcohol as cleaning solvents, hand scrubbing with boiling soap solutions continued to be the most used method for cleaning metals.

In the early nineteen thirties the use of chlorinated solvents for chemical cleaning of metal surfaces became widespread.

The traditional method of laundering fabrics consists of forcing a soap or detergent and water solution through the fibers of the material by hand-scrubbing or some form of mechanical agitation. The development of various solvents for greases and stains in recent years has been of considerable value to the dry-cleaning industry.

The mechanism by which sonic devices clean is not entirely understood at present. It has been experimentally established that little cleaning action takes place unless cavitation is present. Intensities in excess of that required to just produce cavitation at the surface to be cleaned do not seem to produce any better cleaning. Many observers feel that the destructive pressure peaks created by the collapse of microscopic cavitation voids provide the major mechanism by which cleaning is performed. The magnitude of these pressure peaks are estimated to be in the neighborhood of 20,000 psi. Other observers consider that much of the effect is due to secondary chemical actions during the collapse or the occurance of electrical changes of magnitude during the formation and collapse of the cavities. In the cleaning of small parts, cavitation assists in the emulsion of the greases and the solvent and removes small pieces around the edge of holes and cuttings. Even minute cracks or cavities in the metal surface are cleaned out.

The removal of greases from fabrics by sonic devices has been performed experimentally in the laboratory. Ultrasonic frequencies were used on a small sample in a successful test. Cavitation was necessary to produce cleaning. Future industrial use would seem to be

restricted to the cleaning of small areas, because the attenuation of the fabric is very high at ultrasonic frequencies and only that portion of the fabric adjacent to the transducer is cleaned.

The one low frequency application of sonic cleaning was mentioned previously - the domestic washer of the Ultrakust Company.

What are the merits of sonic cleaning? Sonic devices clean much faster than ordinary methods - immersion times of one to ten seconds are normal in the sonic cleaning of metal surfaces. In addition, cavitation provides a degree of cleaning not possible by other methods in some applications.

On the debit side are the high cost of the initial installation and the power required to drive the transducer.

3. THE PROBLEM

The investigation of the application of low frequency sound waves for cleaning purposes was undertaken with a particular problem in mind. A large food processing company is concerned about the large amount of water required with present techniques to wash the vegetables being processed for canning and freezing. The vegetables are presently being washed by high-pressure jets of water being directed on them as they float down a trough about 80 feet long. Thousands of gallons of water per minute are required. As the water requirements for the State of California are increasing much faster than the available supply, the company feels that it may have difficulty in obtaining sufficient water to meet its demands within a few years. Considerable electric power is required to drive the pumps and the trough consumes valuable floor space. If another approach could be worked out for the washing

problem, even though it required a capital investment of the same order of magnitude as is invested in the present system, it would be highly desirable. The environment in which the apparatus would have to work is extremely humid.

For the requirements of the system, it was felt that a mechanically driven transducer would be ideal, if it could be made to produce the desired result with a reasonable degree of efficiency.

# CHAPTER II

## TRANSDUCER DESIGN

As has been earlier stated, the choice of frequency is dictated by the size of object to be cleaned. If the object to be cleaned is large compared to the wavelength ($ka > 1$), there is shadow effect. This results in the far side of the object being shielded from the sound pressure and not being cleaned. If an average size for spinach and broccoli is considered to be about seven inches (17.8 cm), the upper frequency limit is about 1300 cps ($ka = 1$) and the desired working range is about 100 to 150 cps ($ka \ll 1$).

As this device is to operate in the audible range, the annoyance to operating personnel should be considered. As low frequency sounds are less annoying than sounds of the same intensity in the mid-frequency range, this is another reason for the low frequency choice.

The criteria on which the selection of the type of transducer were based were these:

1. the need to establish the principle that vegetables can be cleaned by sonic methods rather than to produce a production device.
2. the need for a sonic device on which the frequency could be varied because of the difficulty of predicting the resonant frequency of the system.
3. the desirability of having a prototype of a device which would operate in a very humid environment and one which would require little specialized knowledge on the part of the future

users for maintenance and repair.

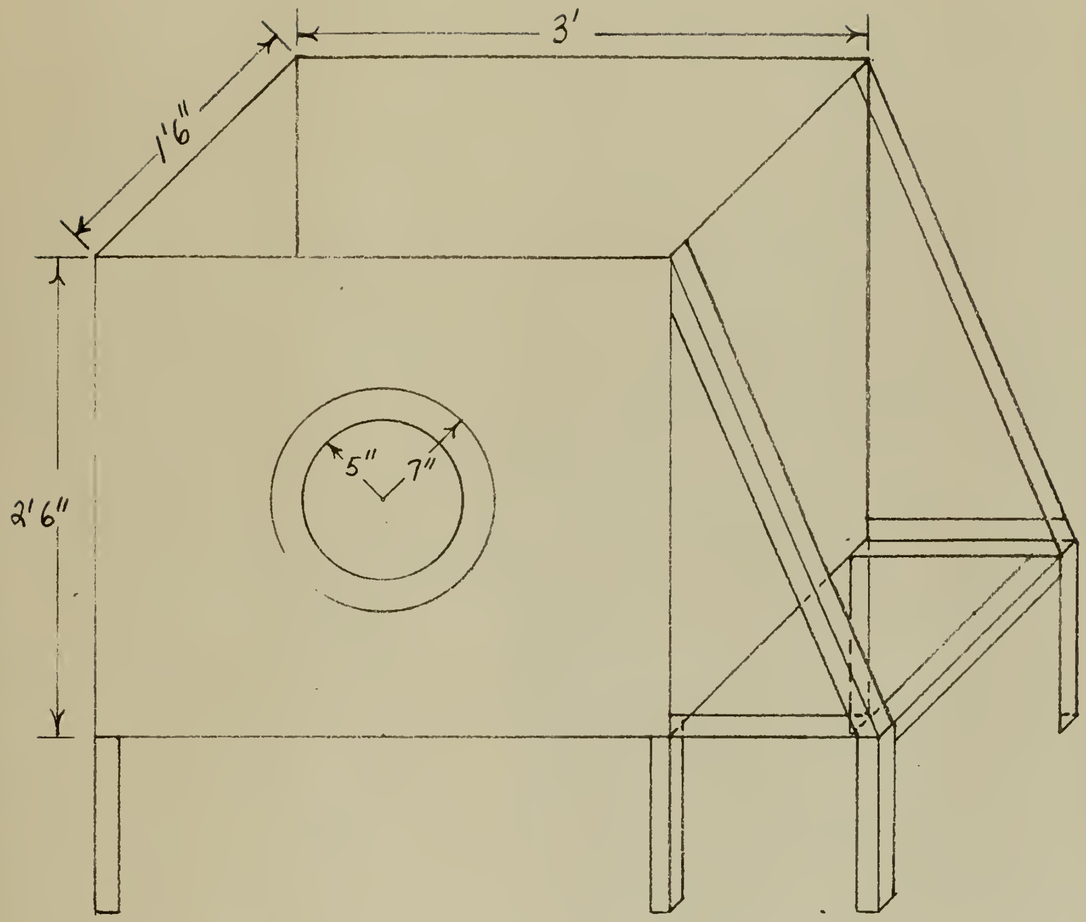
4. the desirability of constructing the device from stock items rather than specially-built items.

The sonic device chosen was an unbalanced force oscillator driving a rigidly clamped diaphragm. This was to be driven by an electric motor through a variable speed drive. A tank was available that was made of 0.100 in. steel plate and was 3 ft. long, $2\frac{1}{2}$ ft. deep, and $1\frac{1}{2}$ ft. wide. A 3/4 hp electric motor was available for the project.

The diaphragm is mounted in the center of one of the longer tank walls (figure 1). The diaphragm periphery is clamped in place by two annular steel rings, one welded to the side of the tank, which are held together by 12 machine screws. A neoprene rubber gasket assists in maintaining a water-tight seal (figure 2).

The diaphragm is driven by a three inch diameter, 12 inch long aluminum cylinder which is attached at one end to the center portion of the diaphragm by six machine screws. The cylinder is supported by two brass bearings which are attached to an angle iron pyramid which is welded to the tank framework (figure 3).

A five inch square, one inch thick block of aluminum is mounted around and attached to the midpart of the cylinder. Two one-half inch axles pass through this block and carry the four weights, the two gears, and the drive pulley (figure 4). The bottom axle is belt driven by a Vickers hydraulic variable speed transmission and the top axle is driven in the opposite direction by the gears. The weights are so oriented that the vertical components of centrifugal force due to the unbalanced mass of the top two weights cancel the vertical components of centrifu-




TANK, SHOWING POSITION OF DIAPHRAGM MOUNT.

FIGURE 1

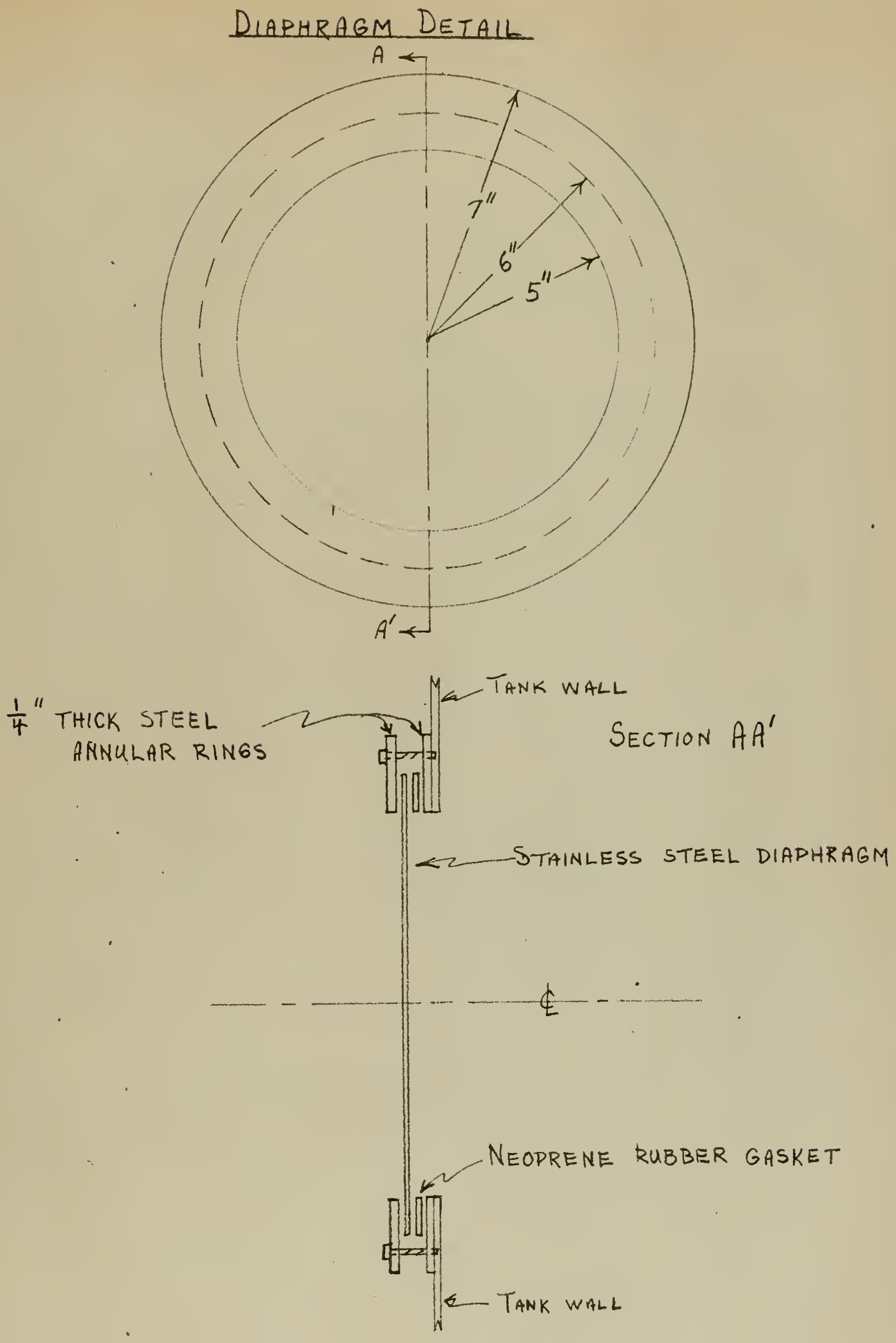

DIAPHRAGM DETAIL
A
7"
6"
5"
A'
TANK WALL
1/4" THICK STEEL ANNULAR RINGS
SECTION AA'
STAINLESS STEEL DIAPHRAGM
℄
NEOPRENE RUBBER GASKET
TANK WALL

FIGURE 2.

## Drive Cylinder

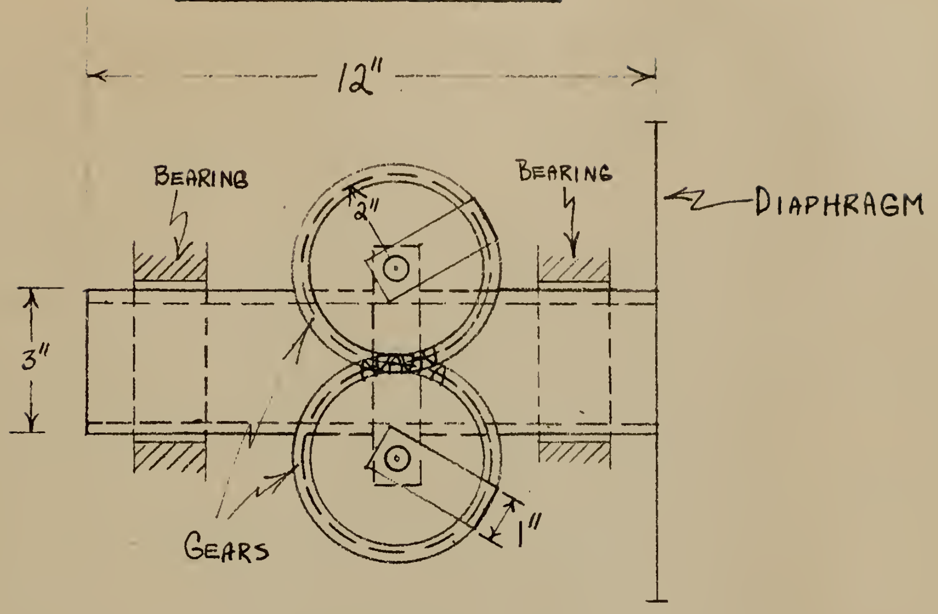



Figure 3

## Drive Block

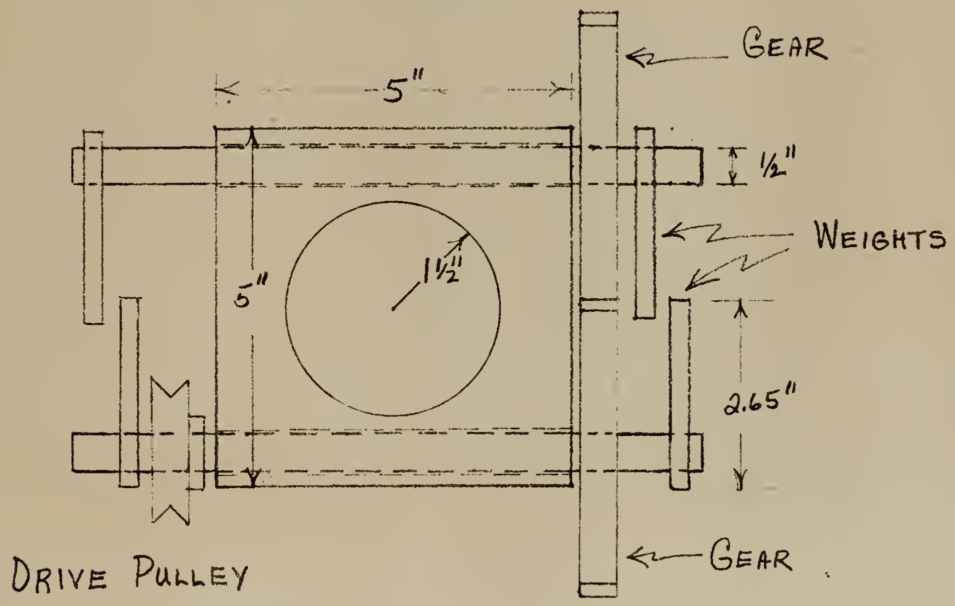



Figure 4

gal force due to the bottom two weights. The horizontal components of centrifugal force from all four weights are additive. The resultant horizontal force is sinusoidal in nature and is transmitted to the diaphragm by the drive cylinder.

The three photographs, figures 5, 6, and 7 show, respectively, an overall view of the tank, detail on the gear side of the drive block, and detail on the pulley side of the drive block. Note that the Vickers variable speed transmission is not included in the photographs.

In attempting to predict the performance of the device, two assumptions were made: (1) the displaced form of the diaphragm, when at its fundamental frequency, will be similar to that of the diaphragm vibrating free and (2) the specific acoustic impedance of the water in the tank will be somewhat near that of an unrestricted body of water.

The first assumption should be reasonably valid, as the diaphragm is to be driven over a three inch circle, rather than just at its center, thus ensuring vibration in a fundamental mode regardless of frequency. The second assumption is less reasonable, as the tank dimensions are small compared to the wavelength and the tank walls may vibrate, changing the loading of the medium on the diaphragm.

The closest simple approximation to the diaphragm is a piston in an infinite baffle. For $\lambda \gg a$ the radiation pattern is effectively non-directional. In this case, the assumption of a sperical wave is reasonably valid.

The specific acoustic impedance of an infinite medium for a spherical sound wave at one radius in front of a piston source of radius a

Figure 5. Sonic Tank

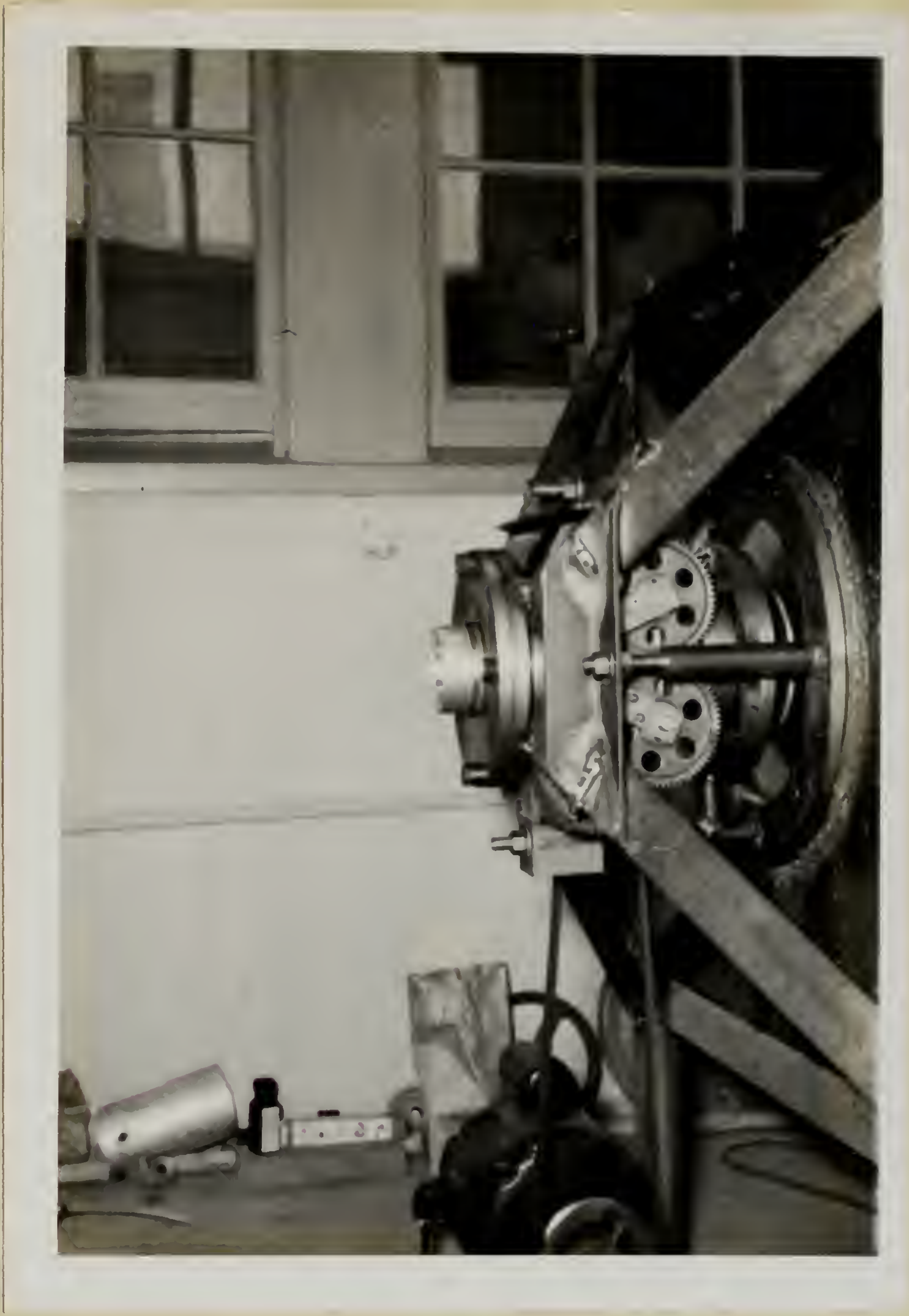


Figure 6. Right side view of drive mechanism.

Figure 7. Left side view of drive mechanism.

is

$$z_a = \rho c \frac{1}{1+\frac{1}{k^2a^2}} + j\rho c \frac{ka}{1+k^2a^2}$$

At f = 60 cps for the diaphragm whose effective radius is 7 cm

$$ka = \frac{2\pi f a}{c} = \frac{2\pi \times 60 \times 7}{1.48 \times 10^5} = .018$$

$$z_a = \rho c \frac{1}{1+\frac{1}{3.2\times 10^{-4}}} + j\rho c \frac{.018}{1+3.2\times 10^{-4}}$$

$$= \rho c(.00032 + j.018)$$

$$\approx j.018\,\rho c$$

In order to produce cavitation in water near the surface, it is necessary that the negative peak sound pressure exceed atmospheric pressure. One atmosphere of pressure equals 1.013 X $10^6$ dynes/$cm^2$. The peak particle displacement in a spherical wave corresponding to this pressure is

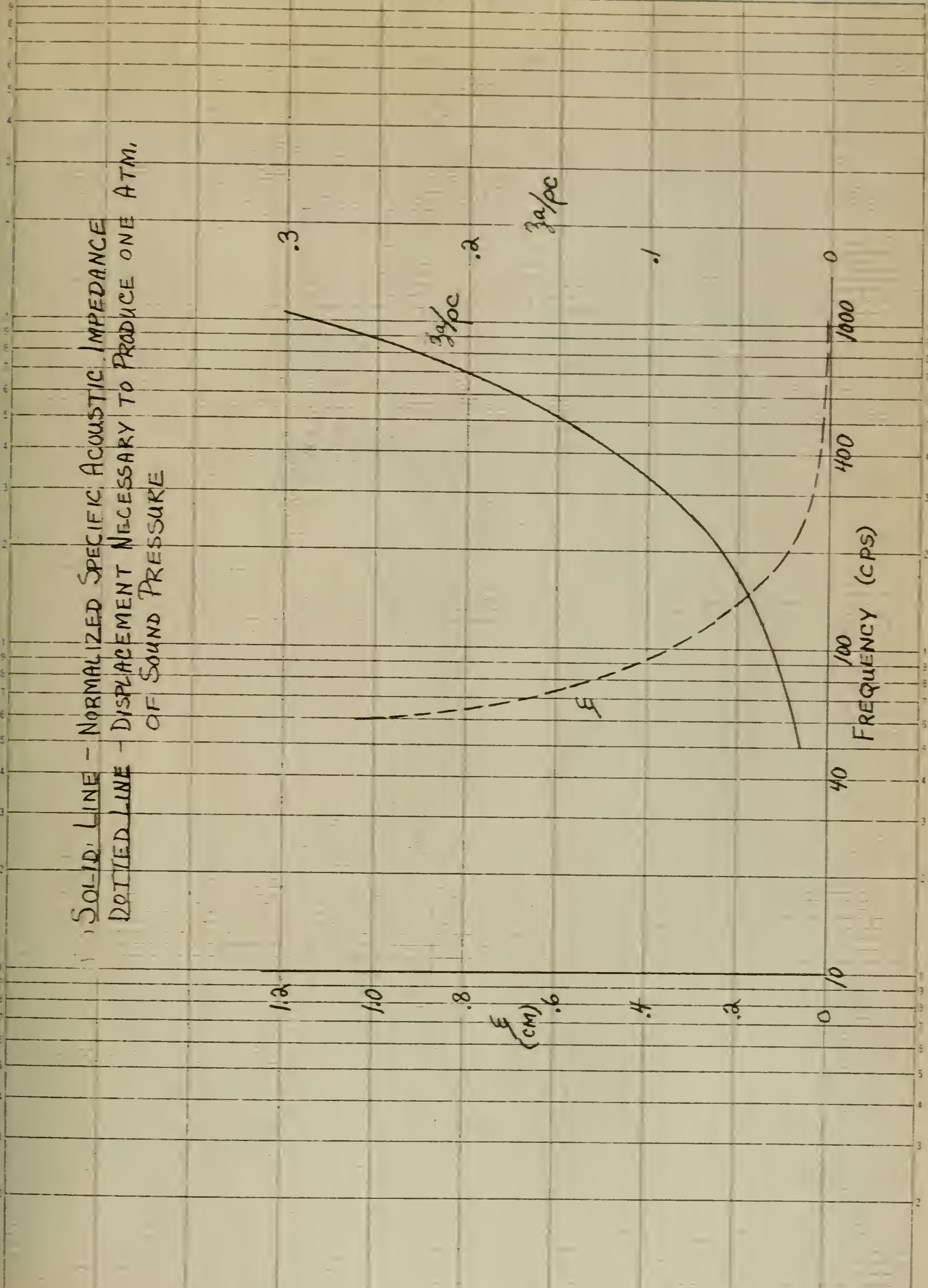
$$\xi = \frac{p}{z_a\omega} = \frac{1.013\times 10^6}{.018\times 1.48\times 10^5 \times 2\pi \times 60} = 1.02 \text{ cm}$$

The normalized specific acoustic impedance and the particle displacement for a sound pressure of one atmosphere for various frequencies are shown in table 1 and are plotted in figure 8.

Table 1

| $f$ | $r_a/\rho c$ | $x_a/\rho c$ | $z_a/\rho c$ | $\xi$ |
|---|---|---|---|---|
| 60 | .00032 | .018 | .018 | 1.02 |
| 120 | .0013 | .036 | .036 | .254 |
| 180 | .0029 | .053 | .053 | .114 |
| 240 | .0051 | .071 | .071 | .064 |
| 500 | .022 | .145 | .147 | .0148 |
| 1000 | .082 | .273 | .285 | .0038 |

FIGURE 8


SOLID LINE - NORMALIZED SPECIFIC ACOUSTIC IMPEDANCE
DOTTED LINE - DISPLACEMENT NECESSARY TO PRODUCE ONE ATM. OF SOUND PRESSURE
za/ρc
ξ
ξ (CM)
1.2
1.0
.8
.6
.4
.2
0
.3
.2
.1
0
FREQUENCY (CPS)
10
40
100
400
1000

The free resonance of a rigidly clamped circular diaphragm is given by

$$f_1 = .47 \frac{t}{a^2} \sqrt{\frac{Y}{\rho(1-\sigma^2)}}$$

For a stainless steel diaphragm of 10-inch diameter .067 inch thickness:

$a = 5'' = 12.7$ cm

$t = .067'' = .17$ cm

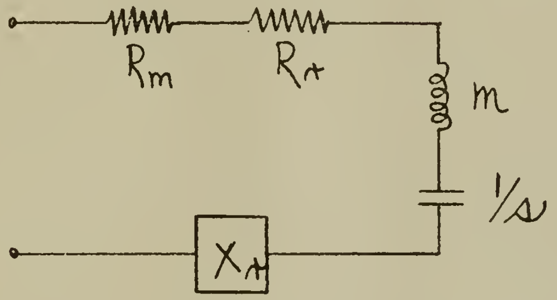
$\sigma = .28$

$Y = 19.5 \times 10^{11}$ dynes/cm$^2$

$\rho = 7.7$ gm/cm$^3$

$$f_1 = .47 \frac{.17}{(12.7)^2} \sqrt{\frac{19.5 \times 10^{11}}{7.7(1-.28^2)}} = 260 \text{ cps}$$

The equivalent circuit of a radiating transducer is




For which the input impedance is

$$Z_{IN} = R_m + R_r + j\,(X_r + \omega m - s/\omega)$$

Resonance occurs at

$$X_r + \omega_o m - s/\omega_o = 0$$

For determining the water load on the diaphragm assume that the effective radius is .55 times the actual radius

$$a = 5 \times .55 \times 2.54 = 7 \text{ cm}$$

For the frequencies of interest $2ka < 1$ and the $X_1(2ka)$ term of the radiation impedance can be approximated by $8ka/3\pi$ and the radiation reactance becomes

$$X_r = \rho c \pi a^2 X_1(2ka) = \rho c \pi a^2 \frac{8 \omega a}{3 \pi c} = \frac{8}{3} \rho \omega a^2$$

Solving for the resonant frequency

$$X_r + \omega_0 m - s/\omega_0 = 0$$

$$\frac{8}{3} \omega_0 \rho a^3 + \omega_0 m = s/\omega_0$$

$$\omega_0 \left( \frac{8}{3} \rho a^3 + m \right) = s/\omega_0$$

$$\omega_0 = \sqrt{\frac{s}{8/3 \rho a^3 + m}}$$

The maximum displacement of a rigidly clamped diaphragm driven over a small area of radius b on one side and balanced by a combination of distributed pressure on the other side plus shear loads at the edge is given by Timoshenko as

$$y = \frac{k_1 F a^2}{Y t^3}$$

where

$k_1 = .103$, a constant

This expression can be expected to hold within the region where the displacement has a linear relationship to the displacing force.

Transforming the equation to the form F/y provides an expression for the stiffness of the diaphragm. This is the only stiffness in the system as the driving cylinder, weights, axles, etc., contribute only mass (plus losses in the bearings).

$$s = \frac{F}{y} = \frac{Yt^3}{k_1 a^2} = \frac{19.5 \times 10^{11} (.17)^3}{.103 \, (12.7)^2} = 5.76 \times 10^8 \text{ DYNES/CM}^2$$

Assuming that the combined mass of the driving system plus the effective mass of the diaphragm to be about five pounds (2270 gms)

$$f_1 = \frac{1}{2\pi}\sqrt{\frac{s}{\frac{8}{3}\rho a^3 + m}} = \frac{1}{2\pi}\sqrt{\frac{5.76 \times 10^8}{\frac{8 \times 1 (7)^3}{3} + 2270}} = 67.6 \text{ cps}$$

As can be seen by table 1 at low frequencies (up to 500 cps), the specific acoustic impedance is reactive and is equal to

$$z_a = \rho c k a = \rho \omega a$$

and

$$\xi = \frac{P}{z_a \omega} = \frac{P}{\rho \omega^2 a}$$

The acceleration corresponding to given displacement is

$$A = \omega^2 \xi = \frac{\omega^2 P}{\rho \omega^2 a} = \frac{P}{\rho a}$$

The force necessary to produce this acceleration of the drive cylinder is

$$F = mA = \frac{mP}{\rho a} = \frac{2.27 \times 10^3 \times 1.013 \times 10^6}{1 \times 7 \times 10^5} = 3200 \text{ NEWTONS}$$

Throughout the low frequencies, then, the force to be supplied to the

diaphragm to produce one atmosphere of sound pressure remains constant.

The effective unbalanced mass of an individual weight, as used on this device, is 49.1 gms at a distance of 3.36 cm from the axle center. The centrifugal force due to a single weight is

$$F_D = m_{EFF}\,\omega^2 r = 49.1 \times 3.36 \times 4\pi^2 f^2 = 6220\, f^2 \text{ DYNES}$$

For four weights

$$F_D = 4 \times 6220\, f^2 = 24900\, f^2 \text{ DYNES} = .249\, f^2 \text{ NEWTONS}$$

A plot of this function is shown in figure 9. It is apparent from this plot that, with the weights mentioned in the preceding paragraph, sufficient driving force will not be produced until 113 cps is reached.

The average power to be supplied to the system is equal to the product of the driving force, the resulting velocity, and the phase angle between them.

$$P_{AVE} = \frac{F u}{2}\cos\theta = \frac{F\omega\xi}{2}\cos\theta$$

$$= \pi f F \xi \cos\theta$$

$$= 10^9 f \xi \cos\theta$$

The function $P_{ave}/\cos\theta$ is plotted in figure 10. The values plotted represent the maximum power required at various frequencies to produce one atmosphere of sound pressure. The real power to be supplied will be reduced by the factor $\cos\theta$ which is difficult to predict.

FIGURE 9

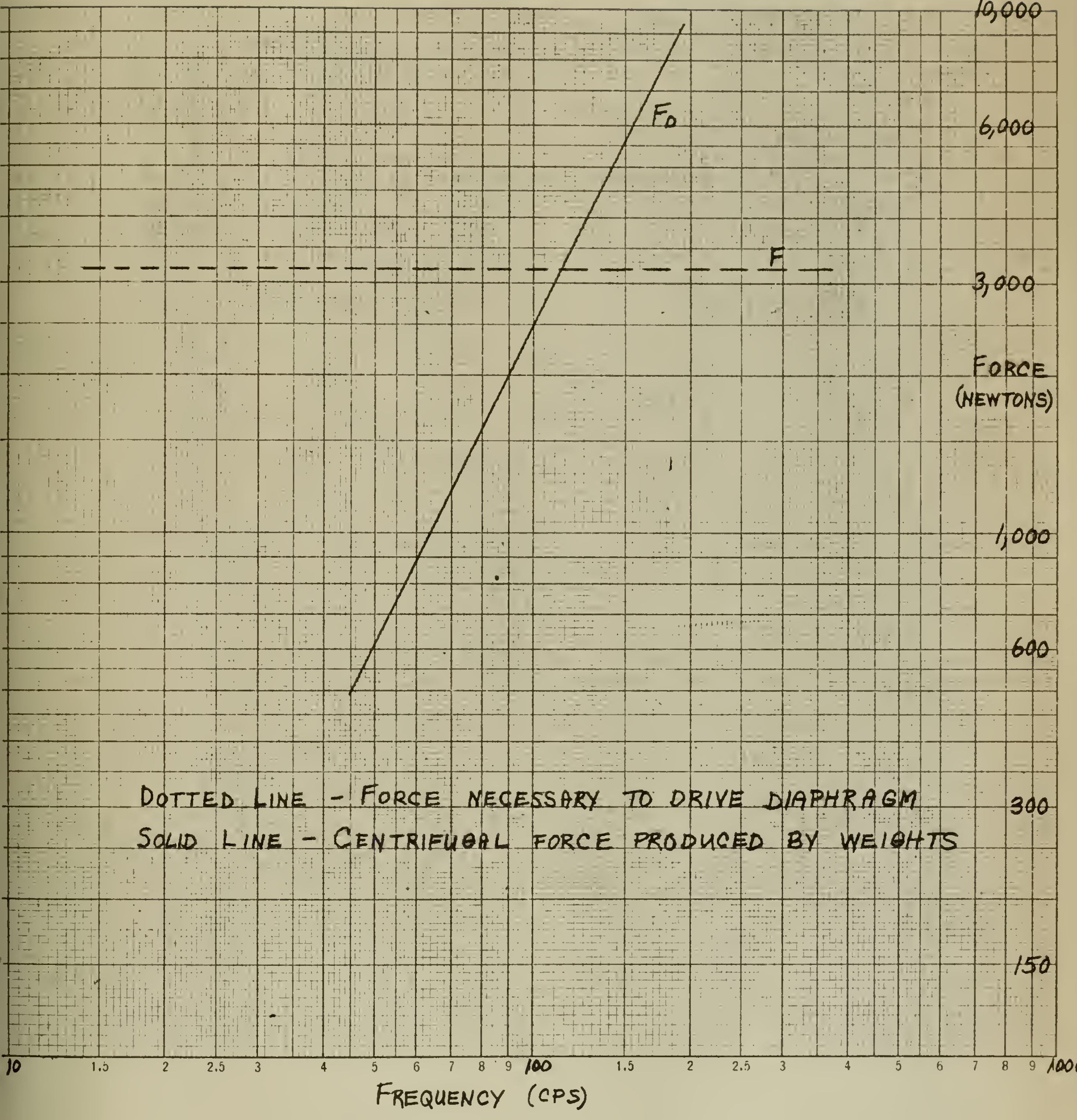
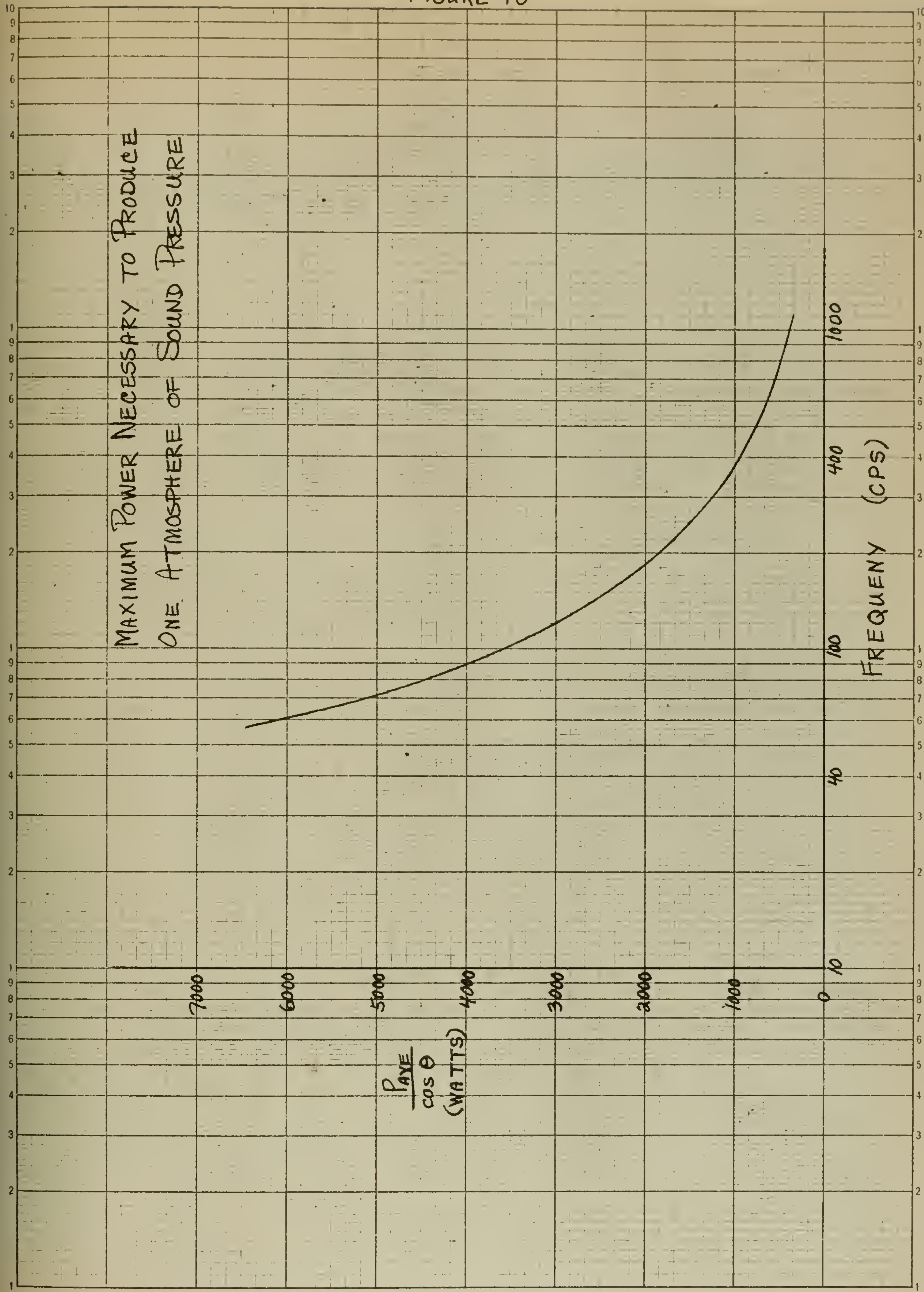

10,000
6,000
3,000
1,000
600
300
150
FORCE
(NEWTONS)
$F_D$
F
DOTTED LINE - FORCE NECESSARY TO DRIVE DIAPHRAGM
SOLID LINE - CENTRIFUGAL FORCE PRODUCED BY WEIGHTS
10
1.5
2
2.5
3
4
5
6
7
8
9
100
1.5
2
2.5
3
4
5
6
7
8
9
1000
FREQUENCY (CPS)

FIGURE 10


MAXIMUM POWER NECESSARY TO PRODUCE
ONE ATMOSPHERE OF SOUND PRESSURE
FREQUENY (CPS)
10
40
100
400
1000
$\frac{P_{AVE}}{\cos\theta}$ (WATTS)
0
1000
2000
3000
4000
5000
6000
7000

# CHAPTER III

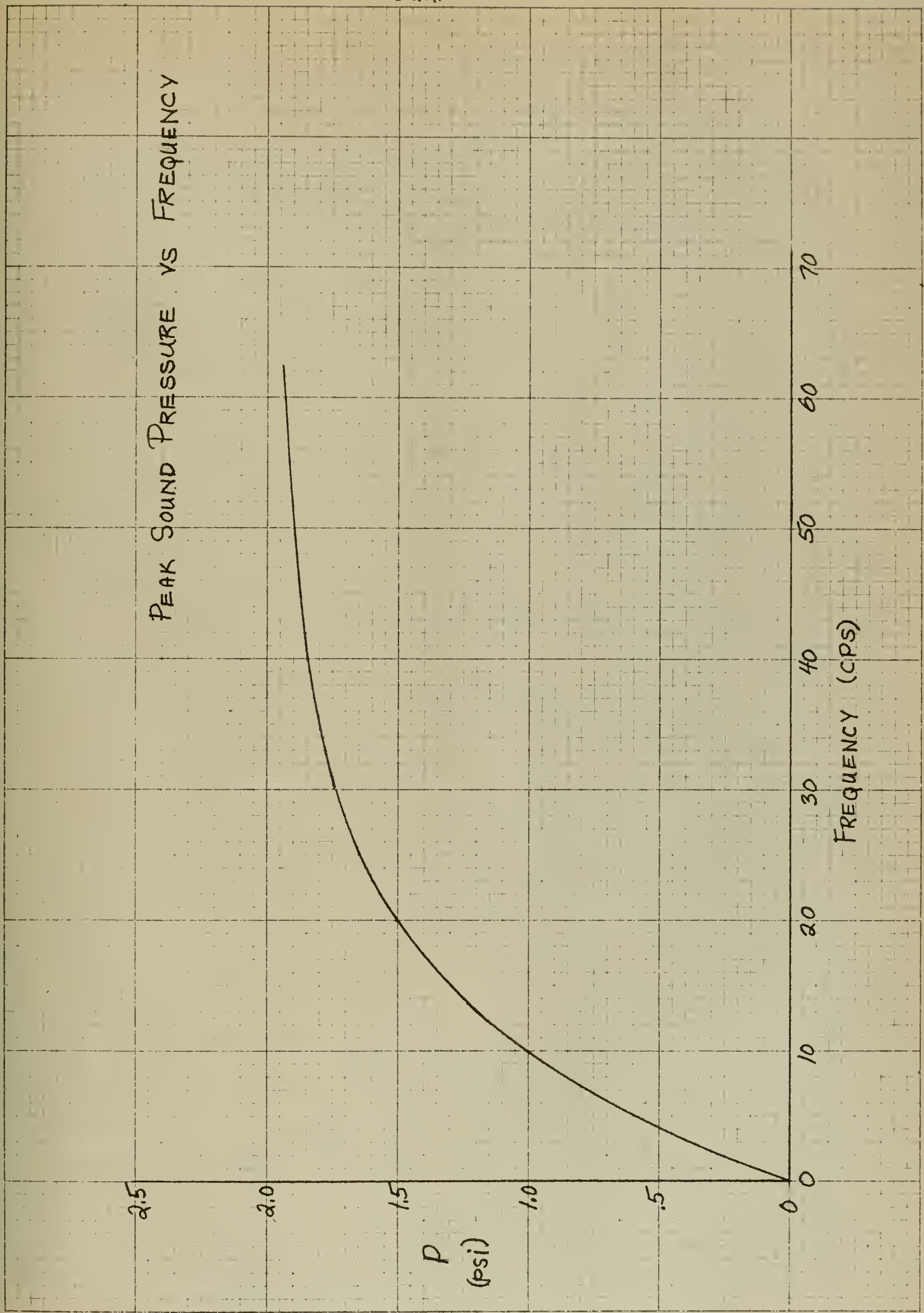
## EXPERIMENTAL RESULTS

Upon completion of the device, the .037" thick diaphragm was installed and initial tests were run for the purpose of determining the existence of any resonant properties of the system. The frequency was varied from zero to the maximum attainable by varying the position of the speed control lever on the Vickers hydraulic drive. The maximum frequency using this diaphragm was 62 cps. No indications of a resonant condition were found. The peak sound pressure varied as shown in figure 11.

Sound pressure in front of the diaphragm was measured with an Endevco Model 2500 pressure pickup mounted in the end of an 18" long, $1\frac{1}{2}$" diameter brass tube. The probe was placed about one radius from the diaphragm on its axis. As $a \ll \lambda$ there is no near field (Fresnel diffraction) effect, and measurements taken close to the diaphragm can be considered valid. The sound pressure field in the tank was explored with the probe and the sound pressure in the center of the tank was found to be much less than that measured close to the diaphragm. The sound pressure immediately adjacent to all of the tank walls was greater than the pressure in the center of the tank (due to the vibration of the tank walls) but was still much less than that close to the diaphragm.

The output of the probe was read directly with an HP 400C vacuum tube voltmeter. The waveform of the sound waves in the tank were observed on an oscilloscope as being essentially sinusoidal. The Endevco

FIGURE 11


PEAK SOUND PRESSURE VS FREQUENCY
P
(psi)
2.5
2.0
1.5
1.0
.5
0
0
10
20
30
40
50
60
70
FREQUENCY (CPS)

pickup is, in effect, a 500 mmfd. capacitance and, consequently, has a high internal impedance at low frequencies. A correction was applied to the probe output to account for the loading of the pickup by the VTVM. The HP 400C is an average reading meter calibrated to read in rms volts. Therefore, all readings were corrected to peak values. The voltage readings were translated into sound pressure readings by use of the manufacturer's calibration curve.

Two thicker diaphragms, .067" and .078", were installed and tested to investigate the effect of greater stiffness in the system. The peak pressure vs. frequency curves were essentially the same as the curve for the .037" diaphragm, except that the maximum attainable frequency was found to be lower as diaphragm thickness was increased. The maximum frequency for the .078" diaphragm was 36 cps.

A diaphragm .020" thick was installed with the hope that a greater amplitude of diaphragm displacement would result. The increase in displacement amplitude was slight and it, too, gave essentially the same results as the .037" diaphragm.

The maximum displacement amplitude of the diaphragms varied from $2\frac{1}{2}$ mm for the .020" diaphragm to 1.25 mm for the .078" diaphragm. Timoshenko's formula,

$$y = \frac{k_1 F a^2}{Y t^3}$$

shows that for a given driving force, the displacement, y, varies inversely with the cube of the diaphragm thickness, t. Also, the maximum attainable frequency with the thicker diaphragms was less than that of the thinner ones, so that the driving force at maximum frequen-

cy was less for the thick diaphragms. The only apparent reason for this decrease in maximum attainable frequency is that the losses connected with driving the thicker diaphragms was greater.

Although the frequency range attainable was fairly limited, the device was evidently operating in the mass controlled region. The predicted resonant frequency of 67.6 cps was based on an estimate of five pounds for the combined diaphragm and drive system mass. The mass of the drive system as actually constructed combined with the effective diaphragm mass was found to be approximately 10 pounds. Computations using this figure result in a resonant frequency of 51.7 cps.

The design computations showed that a peak displacement of one cm is needed to produce an atmosphere of pressure at 60 cps. At 60 cps the actual peak displacement was .2 cm. Applying the spherical wave theory

$$z_a = j\,.018\,\rho c$$

$$P = z_a \omega \xi = .018 \times 1.48 \times 10^5 \times 2\pi \times 60 \times .2$$

$$= 2 \times 10^5 \text{ dynes/cm}^2 = 2.9 \text{ psi}$$

The measured value of P at 60 cps was 2.1 psi. The correlation between this value and the theory is quite good, considering that the theory assumed an infinite medium which patently is not the case. However, it must be noted that $z_a$ is almost completely reactive, which means that little or no real acoustic power is radiated and the sound pressure field is reactive. This also accounts for the very rapid

decrease of sound as the probe was moved from a position close to the diaphragm to the center of the tank. In other words, the sound energy radiated was not sufficient to cause the lack of an infinite medium to effect the system.

It was noted that as the frequency increased, the sound pressure increased rapidly to about two psi and then remained essentially constant. As the frequency increases, the driving force increases as the square of the frequency. However, the peak displacement of the diaphragm decreased with increasing frequency. The net result was that one effect tended to offset the other, so that the total acoustic energy produced remained essentially constant. This is to be expected, as the system was being driven by a device whose output was essentially constant.

An attempt to achieve higher frequency by using a step-up pulley arrangement between the motor and the Vickers unit, so that it was driven at 3450 rpm rather than 1725 rpm, resulted in no greater output. As the speed control on the Vickers was moved from the zero speed position, the frequency would increase to a maximum of 36 cps. Further movement of the control towards the increase speed position resulted in a decrease of oscillation frequency. This was caused by the motor speed decreasing as the overload condition on the motor was reached. Under this method of operation, it was apparent that much of the losses in the system were being caused by the Vickers hydraulic drive as it became quite warm to the touch.

The basic difficulty in attaining high pressure in the tank results from the low value of specific acoustic impedance for a low

frequency spherical wave. The value of $z_a$ can be raised by the use of a closed container. This method has serious drawbacks in a practical system of the type desired.

The size of the diaphragm is important when attempting to produce sound pressure in the medium. If the diaphragm is large compared to the wavelength of the sound corresponding to the frequency with which the diaphragm is moved, the medium is compressed periodically by the diaphragm. When the diaphragm is small compared with the wavelength, something quite different happens. As the diaphragm moves forward, the fluid medium simply slips off to one side without being appreciably compressed. Consequently, only a small amount of vibratory energy is transferred to the medium as sound energy. This was shown mathematically in solving for $z_a$ where it turned out that the real portion of $z_a$ was negligible. Of course, if the small diaphragm acts on a fluid which is confined, as by a cylinder, it, too, will be able to produce compression.

A cylinder of .100" steel plate 10" in diameter and 6" long was welded onto the inside tank wall over the diaphragm (see figure 12). For mechanical reasons, the cylinder was not welded flush against the tank wall but was spot welded in five places with about 1/8" clearance remaining between the tank wall and the end of the cylinder in between the welds. This space was calked with oakum resulting in an estimated 90% water seal.

As no resonant condition for the system had previously been found, and because of the losses in the Vickers drive, the drive was removed and the system was driven directly by the 3/4 hp motor through a

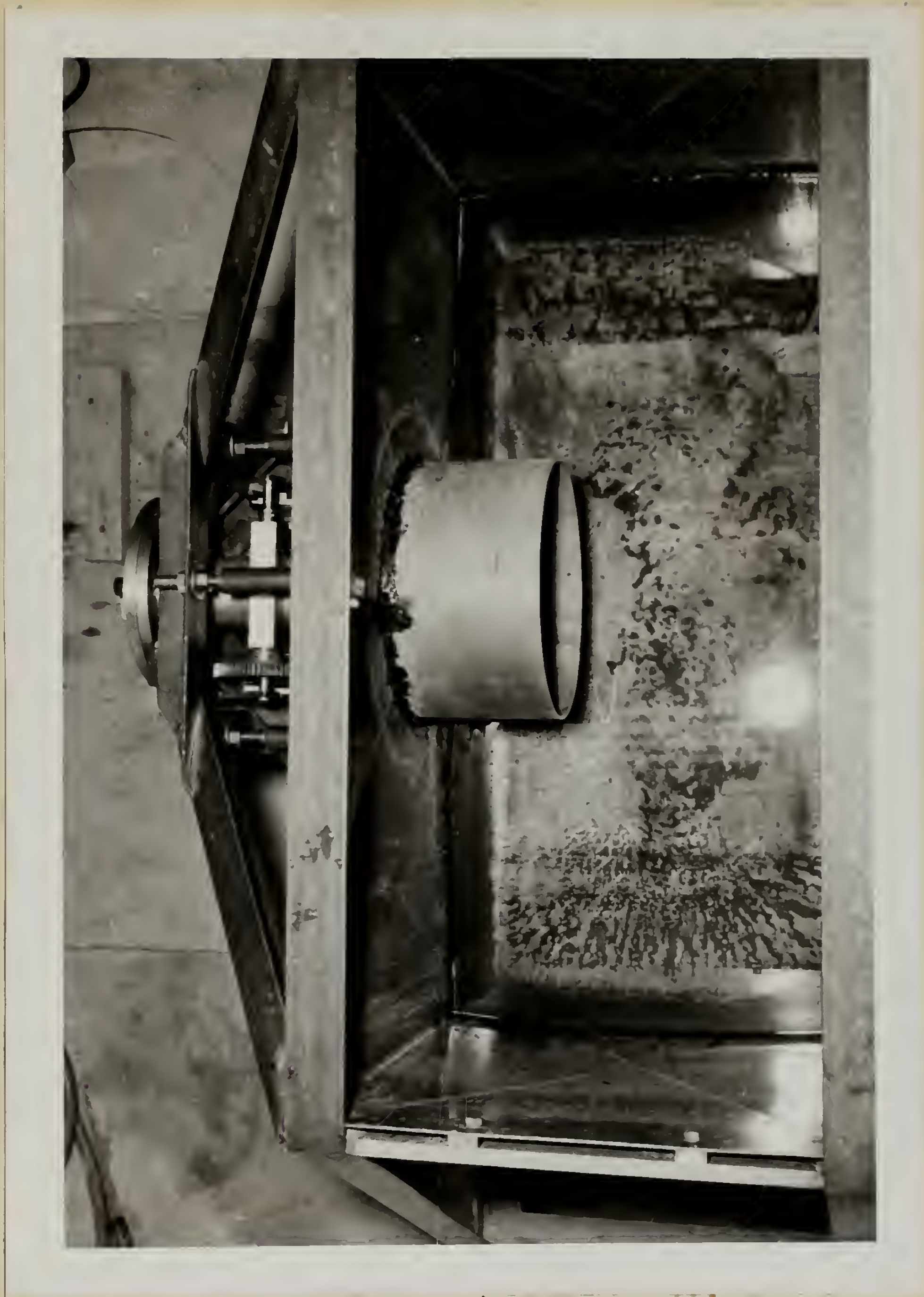


Figure 12. Top view of tank showing the loading pipe in place.

4.5/ 2 pulley ratio. Experimental results with the pipe in place gave a measured 7.07 psi peak pressure at 64.7 cps and 2.5 mm peak displacement. This pressure was measured within the pipe on the axis of the diaphragm and about one radius in front of it. The peak sound pressure measured just inside the open end of the pipe was approximately equal to that measured in front of the diaphragm without the pipe in place. The sound pressure outside the pipe decreased rapidly to a very small value.

The effect of the pipe on the diaphragm loading can be computed by considering the pipe as an acoustic transmission line of characteristic acoustic impedance $\rho c/S$ terminated in an acoustic impedance $Z_l$. At such low frequencies that $2ka \ll 1$ the acoustic impedance of an unflanged pipe is approximately equal to

$$Z_l = \frac{\rho c}{S}\left(\frac{k^2a^2}{4} + .6jka\right)$$

$$ka = \frac{2\pi \times 65 \times 12.7}{1.48 \times 10^5} = .035$$

$$Z_l = \frac{1.48 \times 10^5}{(12.7)^2 \pi}\left(.0031 + j.021\right)$$

$$= j\,6.14$$

The input impedance of the pipe of length l at the end where the diaphragm is located is

$$Z_0 = \frac{\rho c}{S}\,\frac{Z_l + j\frac{\rho c}{S}\tan kl}{\frac{\rho c}{S} + jZ_l \tan kl}$$

$$\frac{\rho c}{S} = \frac{1.48 \times 10^5}{(12.7)^2 \pi} = 292$$

$$\tan kl = \tan .042 = .04$$

$$Z_o = 292 \frac{j6.14 + j292 \times .04}{292 - 6.14 \times .04}$$

$$= j17.9 \text{ g/cm}^4\text{sec}$$

The specific acoustic impedance seen by the diaphragm is

$$z_a = SZ_o = (12.7)^2 \pi \times j17.9 = j9050 \text{ g/cm}^2\text{sec}$$

The peak displacement at the center of diaphragm was measured at .25 cm. The diaphragm can be replaced by an equivalent flat piston whose displacement is .309 times the displacement of the diaphragm center.

$$\xi = .309 \times .25 = .077 \text{ cm}$$

The sound pressure is

$$P = z_a \omega \xi$$

$$= 9.05 \times 10^3 \times 2\pi \times 65 \times .077$$

$$= 2.86 \times 10^5 \text{ dynes/cm}^2 = 4.15 \text{ psi}$$

This theory indicates that an increase in sound pressure immediately in front of the diaphragm should be expected. The value

computed is somewhat less than the measured value (7.07 psi). This is most likely attributable to inaccuracy in the approximation for the displacement of the equivalent piston. The .309 approximation is based on a diaphragm vibrating in its fundamental mode, whereas the actual diaphragm is driven over a three-inch diameter circle which would indicate an average displacement greater than .309 times the center displacement.

Several samples of fresh spinach and broccoli were covered with light mud which was permitted to dry. These samples were then held, one at a time, within the cylinder and about three inches in front of the diaphragm for 10 seconds. A cleaning action was definitely indicated. Some aphis were noted to be within the broccoli heads before washing and most of these were removed by the 10 seconds of sonic washing. The degree of cleaning was considered to be about the same as that produced by the use of high pressure streams of water.

An experimental model of a Bendix sonic cleaner was available that operated at 10 kc (see figures 13 and 14). The bath consisted of a small tank 14" X 19" with a depth of 13 3/8" at the center. The transducer was composed of 24 magnetostrictive elements with 3" X 3" brass plates mounted on the radiating faces and arranged in a semicircular fashion in the bottom of the tank so that some focusing action was present. The 10 kc motor-generator set in a Tocco induction furnace was used to drive the Bendix.

The Endevco pressure pickup probe mounting proved not to be waterproof under the high acoustic pressures present in the Bendix, so another probe with a ½" diameter, 1/8" thick barium titanate disk as

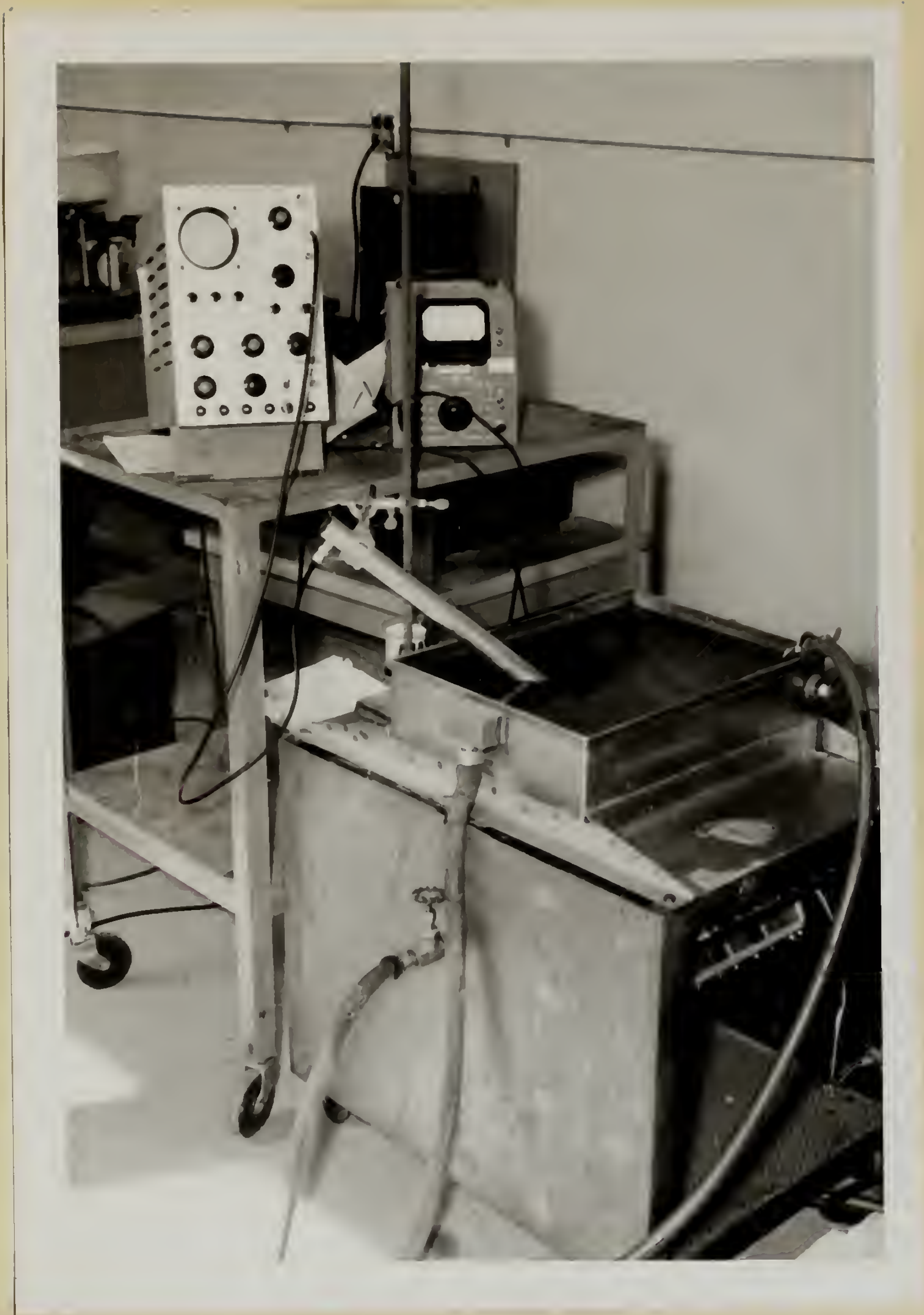


Figure 13. Oblique View of Bendix 10 kc Sonic Cleaner.

Figure 14. Top View of Bendix 10 kc Sonic Cleaner.

the sensitive element was fabricated and calibrated in an air burst chamber. The waveforms of the sound waves in the Bendix tank were observed on an oscilloscope and found to very irregular. A Textronix model 515 scope with built-in voltage calibration was used to measure the peak amplitude of probe output waveforms. This measurement (in volts) was applied to the calibration curve to get the sound pressure.

Peak pressures from 2.7 psi to 19 psi were measured at the focal point of the tank as the output control of the motor-generator set was moved from its minimum setting to its maximum setting. As the sound pressure reached atmospheric pressure at the measuring point, cavitation was observed to be taking place by the appearance of small oscillating bubbles and streamers of foglike nature in the water with the audible output changing from a pure 10 kc note into broadband noise centered 10 kc.

Cleaning of various plant substances was attempted in the Bendix cleaner. In general, the cleaning of leaf surfaces was not as complete as that in the low frequency tank even though the pressure was 18 psi and cavitation was taking place. Examination of the leaves under a microscope revealed that the dirt was incorporated in a dried gelatinous material which was attached to the leaf. This was not true of the dirt on the spinach that was cleaned in the low frequency tank. It was observed that a certain type leaf took on an injected (water soaked) appearance when placed in the sonic washer. Under the microscope, the leaf surface appeared very smooth with large stoma. Another leaf did not become wet or water-soaked. Under the microscope this leaf proved to have a dense population of leaf hairs which evi-

dently prevented the wetting from taking place.

It was observed in the Bendix cleaner that plant materials with a hard surface were cleaned to a much greater extent than those with a soft surface. A weed with fairly large soft leaves and a hard root coated with dirt was immersed in the bath for 10 seconds. The root was well cleaned but the cleaning action on the leaves was negligible. Further study of this phenomenon is indicated.

It is felt by the writer that the results to date indicate that further study of low frequency sonic cleaning is warranted. Low audio frequencies seem to provide better cleaning than high frequency. A more powerful driver for the unbalanced force oscillator with a good variable speed drive capable of transmitting several horsepower is needed to fully investigate its capabilities. The curves for $z_a$ and driving power required indicate a frequency in the range of 200 to 300 cps would prove to be a better operating frequency than the very low frequencies attainable with the present device. As the rotating weights must rotate at 12,000 rpm to produce a vibratory frequency of 200 cps, this frequency would certainly be the upper limit and the device would require careful mechanical design to be capable of it. It does not appear that the unbalanced force oscillator is the best choice that one could make for a driving system.

It would not be practical to immerse the vegetables to be cleaned into the interior of the loading cylinder in a commercial model, but increasing the diaphragm size, and having a cutout in the cylinder through which the vegetables could pass, should be feasible.

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